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VAN NOSTRAND SCIENCE SERIES.

Worm and Spiral Gearing

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Worm and Spiral Gearing

BY

FREDERICK A. HALSEY,
ASSOCIATE EDITOR "AMERICAN MACHINIST."

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PREFACE.

JUSTIFICATION for the republication of the contents of this book is found in the still prevalent opinion among designers of machinery that worm gearing is necessarily short lived and of low efficiency, and in the fact that the methods of laying out spiral gearing are not as widely understood as the merit and convenience of that form of gearing make desirable.

The theory of worm gearing is well fortified by the collection of facts from experience given herein, and it points out clearly the procedure to be followed in order to insure durability and efficiency. Both analytical and graphical methods of laying out spiral gearing are given, and these, it is believed, will meet the needs and tastes of all.

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It should be mentioned that with spur gears, when stock cutters are used, the diameters and center distances must be such as go with whole numbers of teeth. The property of spiral gears by which any center distance may be accommodated is therefore one which is not shared by spur gears. Should those sections of this book which relate to unchanged center distances be omitted, its remaining solutions would be as complete as is possible with spur gears.

As uncertainty has arisen in the minds of some readers, it should also be mentioned that the solutions here given relate exclusively to gears having their shafts at right angles.

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PART I.

WORM GEARING.

In view of the good results now being obtained with worm gearing, the old prejudice against that form of gearing, on account of its supposed low efficiency and short life, is dying out. These good results are the outcome of the application of principles which are by no means a late discovery, and it is expected that what follows will contain much that to some readers is not new. At the same time it is an undoubted fact that the best practice with worms is understood by but few, relatively speaking, and the corroboration of the theory by examples from practice which follow, is believed to be new. No better illustration of the fact that good practice with worm gearing is not

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yet widely understood could be given than the statement in a recent and excellent work on gearing that "the diameter of the worm is commonly made equal to four or five times the circular pitch," the fact being that such proportions are distinctly bad if the worm is to do hard work.

It should be stated at the beginning that while what follows is not offered as a presentation of all the data necessary for assured success with worms under all conditions, it is hoped to make the general conditions of successful practice plain, and to present the "state of the art" as it exists to-day.

The essential change in practice which has improved the results obtained with worm gearing has been an increase in the pitch angle over what was formerly considered proper. There is no doubt whatever that this change has increased the efficiency of the gear, and, what is of more importance, has reduced the tendency to heat and rapid wear. This is

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not only a fact, but it is a sound conclusion from theoretical considerations, which might have been predicted under proper examination.

THEORY OF WORM EFFICIENCY.

The reason why an increase of pitch, other things being equal, or in other words, an increase of the angle of the thread, gives these results, will be understood from Fig. 1. If *a b* be the axis of the worm and *c d* a line representing a thread, against which a tooth of the wheel bears, it will be seen that if the tooth bears upon the thread by a pressure *P*, that pressure may be resolved into two components, one of which, *e f*, is perpendicular, while the other, *e g*, is parallel to the thread surface. The perpendicular component produces friction between the tooth and the thread. The useful work done during a revolution of the thread is the product of the load *P* and the pitch of the worm, while the work lost in friction is the product of the perpendicular pres-

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sure $e\,f$, the coefficient of friction and the distance traversed in a revolution, which is the length of one turn of the thread. Now, if the angle of the thread be doubled, as indicated, the load P remain-

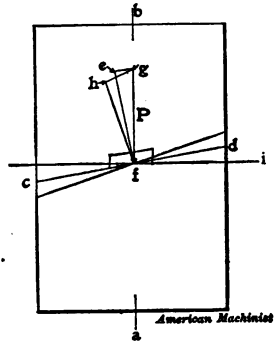

Fig. 1. THE PRINCIPLE OF WORM EFFICIENCY.

ing the same, the new perpendicular component $f\,h$ of P will be slightly reduced from the old value $e\,f$, while the length of a turn of the thread will be slightly increased. Consequently their product and the lost work of friction per revolution

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will not be much changed. The useful work per revolution will, however, be doubled, because, the pitch being doubled, the distance traveled by P in one revolution will be doubled. For a given amount of useful work the amount of work lost is therefore reduced by the increase in the thread angle, and, since the tendency to heat and wear is the immediate result of the lost work, it follows that that tendency is reduced. For small angles of thread the change is very rapid, and continues, though in diminishing degree, until the angle reaches a value not far from 45 degrees, when the conditions change and the lost work increases faster than the useful work, an increase of the angle of the thread beyond that point reducing the efficiency.

This general consideration of the subject shows the principles at the bottom of successful worm design, but a more exact examination is desirable. According to Professor Barr the efficiency of a

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worm gear, the friction of the step being neglected, is :

$$e = \frac{\tan \alpha \,(1 - f \tan \alpha)}{\tan \alpha + f}$$

in which

e = efficiency,

α = angle of thread, being the angle $d\,f\,i$ of Fig. 1,

f = coefficient of friction.

To study the effect of the step, a convenient assumption is that the mean friction radius of the step is equal to that of the worm. This assumption would only be realized in cases where the step is a collar bearing outside the worm shaft, and the preceding and following formulas therefore represent extreme cases, one of a frictionless step, which would be approximated by a ball bearing, and the other of a step having about the extreme friction to be met with. Most actual cases would therefore fall between the two. Again, according to Professor

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Barr, the efficiency of a worm and step on the above assumption is : *

$$e = \frac{\tan \alpha (1 - f \tan \alpha)}{\tan \alpha + 2f} \text{ (approximately)}$$

Notation as before.

These formulas give no clear indication of the manner in which the efficiency varies with the angle, and the diagram, Fig. 2, has been constructed to show this to the eye. The scale at the bottom gives the angles of the thread from 0 to 90 degrees, while the vertical scale gives the calculated efficiencies, the values of which have been obtained from the equations and plotted on the diagram. The upper curve is from the first equation, and gives the efficiencies of the worm thread only; while the lower curve, from the second equation, gives the combined efficiency of the worm and step. In the

* In Professor Barr's formulas it is assumed that the worm thread is square in section. Thread profiles in common use affect the results but little.

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calculations for the diagram it is necessary to assume a value for f, and this has been taken at .05, which is probably a fair mean value. The experiments made by Mr. Wilfred Lewis for Wm. Sellers & Co. showed an increase of efficiency with the speed. The present diagram may be considered as confined to a single speed, and at the same time is not to be understood as showing the exact efficiency to be expected from worms, but rather to exhibit to the eye the general law connecting the angle of the thread with the efficiency.

The curves will be seen to rise to a maximum and then to drop. The exact values of the angle of thread to give maximum efficiency may be easily found by the methods of the calculus, the results being :

For worm thread alone the efficiency is at a maximum when

$$\tan \alpha = \sqrt{1 + f^2} - f.$$

Substituting the value of f (.05) used in calculating the diagram, this becomes

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tan α for maximum efficiency = .9512,

and by referring to a table of natural tangents we find that

α for maximum efficiency = 43° 34.

Similarly for the worm and step the result is

tan α for maximum efficiency = $\sqrt{2 + 4f^2} - 2f$, which for $f = .05$ = 1.318,

and a table of tangents tells us again that

α for maximum efficiency = 52° 49′.

Of more importance than the angle of maximum efficiency is the general character of the curves, of which the most pronounced peculiarity is the extreme flatness, showing that for a wide range of angles the efficiency varies but little. Thus, for the upper curve there is scarcely any choice between 30 and 60 degrees of angle, and but little drop at 20 degrees.

At first sight the lower curve might be

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thought the most useful of the two, as it includes the effect of the step, but a little consideration will show that this is not the case. For most cases in which worms are used the efficiency of the transmission, as such, is of very little account. What the designer concerns himself with is the question of durability and satisfactory working, and the results to be expected in this respect are best shown by the upper curve, in which high efficiency means a durable worm. Throughout this discussion, in fact, the chief significance of efficiency lies in the fact that low efficiency means rapid wear and vice versa.

EXPERIMENTAL CORROBORATION OF THE THEORY.

The experiments of Wm. Sellers & Co., before referred to, go far to confirm the soundness of the above views. From the present standpoint it is unfortunate that those experiments did not cover a wider range of worm thread angles—those act-

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ually used being 5 degrees, 7 degrees, and 10 degrees. Other experiments were, however, made on spiral pinions of higher angles, spiral pinions being understood by Mr. Lewis to mean those pinions having the mating gear a true spur, the pinion shaft being at a suitable angle with the gear shaft to bring the pinion in proper mesh—a construction which is exemplified in the well-known Sellers planer drive. Mr. Lewis gives a formula by which the efficiencies of worms can be calculated from those for spiral pinions, and in the absence of direct experiments on worms of high angles, his results for spiral pinions have been modified by this formula to read for worms. The results for the two forms of gearing differ by less than five per cent. for the extreme case of his experiments. To compare the results obtained by Mr. Lewis with Professor Barr's formula, a speed has been selected from the experiments giving the nearest coefficient of friction to that used in obtaining the curves of Fig. 2. The

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The increase of efficiency with the speed is a valuable property of worms, and enables them to do much more work than they otherwise would. Thus the 20 degree worm at 20 feet per minute lost 21½ per cent. of the work in friction. Increasing the speed to 40 feet doubled the work applied, and, had the efficiency remained constant, would have doubled the friction heat to be dissipated. In point of fact, this increase of speed diminished the percentage of loss to 17, and the amount of loss and heat, instead of being doubled, was only increased in the ratio of 160 to 100. It is plain from the diagram, however, that this action does not continue much beyond a velocity of 200 feet per minute, beyond which the amount of loss must be more nearly proportional to the speed, and this doubtless has some connection with the fact observed by Mr. Lewis that 300 feet per minute is the limit of speed when the gears are loaded to their working strength, and that the best conditions are obtained at

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about 200 feet per minute. It is proper to add, however, that in the cases from practice given later there are three which have been made repeatedly, and which are conspicuously successful, in which the velocity exceeds 600 feet, and one in which it exceeds 800 feet. No doubt, in all such cases, if the pressure on the teeth could be known it would be found to be light.

It will be seen that an increase of speed for any worm under constant pressure leads to an increase of friction work, and the limit is reached when the worm is no longer able to carry off the heat generated fast enough to prevent undue rise in temperature. Furthermore, this limiting speed depends upon the pressure, it being higher for low pressures than for high. A worm having an angle which might be successful at low speed may fail at high speed; but it would seem that any worm which is successful at high speed should also be successful at low, which is in accordance with mechanical instinct.

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There are, it will be observed, two methods of increasing the pitch angle. The diameter may be kept constant and the pitch be increased, or the pitch may be kept constant and the diameter be reduced. From a mathematical standpoint, these two methods are identical; that is, at a given pitch line velocity a worm of a given angle should have the same efficiency, regardless of the diameter; but in a mechanical sense the methods are not identical. The worm of the larger diameter would naturally have a gear of wider face and the pair, having greater area of tooth surface in contact would carry a larger load.

EXAMPLES FROM PRACTICE.

It is impossible to say who was the first to recognize the significance of the pitch angle as a factor in the satisfactory performance of worm gearing, but it may be mentioned as a matter of interest that the exhibit of the Hewes & Phillips Iron Works at the Newark Industrial Exhibi-

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tion of 1873 included several worm-driven planers, in which the worms were double threaded and had a pitch angle of 15° 15′, a pitch diameter of 3½ inches, a pitch of 3 inches, and a speed, cutting, of 256 and backing of 640 r. p. m., which give pitch line velocities of 237 and 590 feet. This worm was successful, and was many times repeated; but later on Hewes & Phillips were struck by the high belt speed idea, and in order to increase the belt speed they changed the worm to 6.16 p. d., 1¾ inches pitch, single thread; speed, cutting, 446, and backing 1,110 r. p. m., giving a pitch angle of 5° 15′ and pitch line velocities of 720 and 1780 feet. This worm was a failure, and was soon changed to 6.16 p. d., 3½ inches pitch, double thread; speed, cutting, 281, and backing 700 r. p. m., giving an angle of 10° 15′ and pitch line velocities of 452 and 1,130 feet. This worm did better than the last, but not so well as the first. By this time the lesson was learned, and Hewes & Phillips set out to use a worm

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of 30 degrees pitch angle. Structural considerations, however, prevented the use of so high an angle and they compromised on 20 degrees, the final worm resulting from this experience having a pitch diameter of 2.63 inches, with 3 inches pitch, quadruple thread, the speed cutting being 300 and backing 700 r. p. m., giving pitch line velocities of 205 and 480 feet, and this remained the standard angle as long as these planers were manufactured. The writer has seen one of these 20 degree worm gears, opened up after twelve years' use, and the wear disclosed was very slight—no shoulder being in existence. As a result of the experience outlined above, this house adopted the standard practice of making the worms as small as possible in diameter, and giving the threads in all cases a pitch angle of 20 degrees. The form of tooth used was the epicycloidal, while the materials used were hard cast-iron for the gear and case-hardened open-hearth steel for the worms.

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These Hewes & Phillips worms are plotted in Fig. 3 as crosses 1, 2, 3, 4, of which 1 is the 15° 15′, 2 the 5° 15′, 3 the 10° 15′, and 4 the 20°, the first and last being successes, and the second and third failures.

In plotting these worms, and all others having pitch line velocities above 200 feet, the crosses are placed near and above the 200 feet curve. It is unfortunate that we have no curves for higher speeds, but Mr. Lewis recommends the use of the 200 feet line for all higher speeds. Leaders connecting different crosses indicate the same worm at different speeds in all cases. The letters *s* and *f* on the diagram mean success or failure in all cases.

Fig. 4 is a drawing of worm 3 (failure) and Fig. 5 shows worm 4 (success), and no more instructive pair of drawings could be imagined than these. The pitches are not far different, and what difference there is is in favor of the larger worm. The duty is the same, the gears are of about the same diameter, and

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the revolutions per minute are nearly the same. The essential change is in the increase of the pitch angle by a reduction of the diameter, and this changed failure to success.

The Newton Machine Tool Works use worm gearing in many of their machines, notably their cold saw cutting-off machines. In the earlier machines of this class the worm had a pitch diameter of $2\frac{1}{8}$ inches, with a pitch of 1 inch, single thread, the revolutions per minute being 765. These figures give a pitch angle of 6° 20′, and a pitch line velocity of 572 feet. This machine could be operated, but not with satisfaction on account of the heating and short life of the worm. The worm was then increased in pitch by making it double threaded, giving a pitch angle of 12° 30′, the speed being reduced to 500 revolutions per minute, giving a pitch line velocity of 375 feet. The change proved to be a great improvement, heavier work than was before possible being done after the change without

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distress or difficulty, and this worm has since been applied to a large number of machines with entire success. A still later worm used on these machines has a pitch diameter of $3\frac{7}{8}$ inches and a pitch of 4 inches. triple threads, giving a pitch angle of 18° 15′, and this is found to be a still further improvement. This last worm is used on a wide variety of machines and at a variety of speeds from 40 to 680 r. p. m., giving pitch line velocities of from 40 to 685 feet, and with uniformly good results. In many cases it is used without an oil cellar, though for comparatively light work. The form of thread used is the involute, and the material is hardened steel for the worm and bronze for the wheel. These Newton worms appear in Fig. 3 as 5, 6, 7, of which 5 is nearly a failure, while 6 and 7 are entirely successful. The second Newton worm—the one appearing in Fig. 3 as 6—is shown in Fig. 6.

Another habitual user of worms is John Bertram & Sons, of Dundas, Ontario, Can-

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ada, who employ them in all their planers, and use largely a worm of 3.18 inches pitch diameter, 4 inches pitch, quadruple threads, the speed, cutting, being 186 and reversing 744 r. p. m. These figures give a pitch angle of 22 degrees, and pitch line velocities of 155 and 620 feet. This worm appears in Fig. 3 as 8, the vertical position for the higher speed being again uncertain. These worms are highly successful, as the writer knows from repeated observation. Both worm and wheel are of cast-iron, the thread being Brown & Sharpe standard. The Bertram worm is shown in Fig. 7. In reading this drawing it should be remembered that the conventional representation of a worm, with the threads shown by straight lines, shows a larger apparent pitch angle than the true one, as shown by a true projection.

Another case of failure was a worm drive applied to a large boring machine, the worm being 12 inches pitch diameter, 8 inches pitch, quadruple thread, speed

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80 r. p. m. and above, worm of forged steel, wheel of bronze, oil cellar lubrication. These figures give a pitch angle of 12 degrees and a pitch line velocity of 250 feet. This worm is located on Fig. 3 as 9.

Still other cases of change from failure to success are supplied by Mr. Jas. Christie, of the Pencoyd Iron Works. The first of these relates to a boring machine, which was, by the makers, supplied with a worm drive having a worm of $5\frac{1}{2}$ inches pitch diameter, $1\frac{1}{2}$ inches pitch, single thread, steel worm and cast-iron wheel, average speed 150 r. p. m. These figures give a pitch angle of 5 degrees and a pitch line velocity of 215 feet. This was a failure, but was successfully replaced by a worm of $4\frac{7}{8}$ inches pitch diameter, $2\frac{1}{2}$ inches pitch, and the same number of revolutions, which figures give a pitch angle of 9° 15′ and a pitch line speed of 190 feet. These two worms appear as 10 and 11. This successful worm lies in the region of unsuccessful ones, but the in-

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fluence of the increased lead angle is unmistakable. The fact of its success is probably due to the pressure on the teeth being well below the working strength, or to the speed being moderate, or both.

The second case, of which the data were supplied by Mr. Christie, relates to two heavy milling machines, in which the cutter spindles were driven by worms 6 inches pitch diameter by 1½ inches pitch, single thread. It was found that the cutters could be run much faster than was originally contemplated, and the worms were consequently speeded up to about 500 r. p. m. In these machines cast-iron worm wheels were speedily destroyed, while hardened steel worms and bronze wheels would last about a year. Later two more machines were built having steel worms and bronze wheels, the worms being 4½ inches pitch diameter by 5 inches pitch, quadruple threads, speed 280 r. p. m. These worms have been in use six years, and are described as being "good as new." The data given for the first

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worm give a pitch angle of 4° 30′ and a pitch line velocity of 785 feet. It appears in Fig. 3 as 12. The pitch angle of the second worm is 19° 30′, and its pitch line velocity 328 feet. It appears in Fig. 3 as 13.

Mr. Christie has made many successful changes, of which these are typical, and he now uses worms with great freedom and success. His general conclusion is that good worms begin with those having the pitch about equal to the diameter, giving a pitch angle of 17° 15′.

Another equally striking case of success accompanying an increase of the pitch angle is supplied by Mr. W. P. Hunt, of Moline, Ill., who says :

"In building a special double-spindle lathe I wished to use a worm drive, and having a single-thread ¾-inch pitch hob, 2¾-inch outside diameter, I decided to work to that, and made my gear with twenty-six teeth, giving a speed reduction of 26 to 1. The worm was to run at 460 revolutions per minute, but upon starting

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the machine I found it impossible to keep the worm and gear cool, and the belts would not pull the cut.

"Accordingly I decided to make a new worm and hob having the same outside diameter as the one first tried, but with double thread and 2-inch pitch, and a new gear having forty-eight teeth, giving me a speed reduction of 24 to 1, or less than at first.

"Upon starting the machine with the new worm and gear, not only did it run perfectly cool, but the belts have ample power. We use graphite and oil on the worm, and it is not enclosed."

Mr. Hunt does not give the pitch diameter of his worms, but assuming the threads to have been in accordance with the Acme standard, the pitch diameters are $2\frac{3}{8}$ and $2\frac{1}{4}$ inches respectively, the thread angles being 5° 44′ and 15° 48′, and the pitch line speeds 286 and 271 feet per minute. Mr. Hunt's worms are plotted in Fig. 3 as 17 and 18.

Three other cases of successful worms

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under heavy duty are found in milling machines which have been repeated many times. The first two worms would ordinarily be described as spiral gears. The shafts are at right angles and the action is that of worms, and they are properly included here.

The first of these, which appears as 14 in the diagram, has a pitch diameter of 2¼ inches, a pitch angle of 45°, and a speed varying between 180 and 945 r. p. m., giving pitch line velocities of 106 to 555 feet per minute. Both gears are of cast-iron. The second, 15 in the diagram, is of the same style, and has the same pitch diameter, with speeds varying between 90 and 472 r. p. m., giving pitch line velocities of 53 to 277 feet per minute. The third, 16 in the diagram, is a true worm, 2¼ inches pitch diameter, lead 1.333, triple thread, speed 200 to 1,442 r. p. m., bronze wheel and hardened steel worm. These figures give a pitch angle of 10° 45′, and a pitch line velocity of 118 to 845 feet per minute. While this

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worm is entirely successful, it was at first a failure, and was made successful only by careful attention to the materials used.

LIMITING SPEEDS AND PRESSURES.

A very important point connected with worm design, and one on which data are very scarce, is the limiting pressures for various speeds at which cutting begins. The paper by Mr. Lewis contains some information on this subject, and the accompanying table supplied by Mr. Christie, from experiments made by him, supplies the most definite additional data on the subject known to the writer. In all cases the worms were of hardened steel and the worm wheels of cast-iron. Lubrication by an oil bath.

There is real need of a comprehensive series of experiments on this subject. It is obvious enough that a worm, otherwise well designed, might fail from having too high a speed for its load. Were such data at hand it would seem that with existing knowledge of the influence of

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LIMITING SPEEDS AND PRESSURES OF WORM GEARING.

	Single Thread Worm 1″ Pitch 2⅞ Pitch Diameter				Double Thread Worm 2″ Pitch 2⅞ Pitch Diameter			Double Thread Worm 3½″ Pitch 4½ Pitch Diameter		
Revolutions per minute	128	201	272	425	128	201	272	201	272	425
Velocity at pitch line in feet per minute.....	96	150	205	320	96	150	205	235	319	496
Limiting pressure in pounds.....	1,700	1,300	1,100	700	1,100	1,100	1,100	1,100	700	400

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the angle of the thread, worm design might be made a matter of comparative certainty. Especially should the behavior of worms at speeds above 200 feet per minute be subjected to further experiment, as it is frequently necessary to use speeds above that figure, and there can be no doubt that higher speeds are entirely feasible if suitable pressures accompany them. The speed as a factor should be kept in mind equally with the pitch angle. A worm may fail because of too high a pitch line velocity as well as because of too low a pitch angle.

The number of cases cited is too few for certainty in drawing general conclusions, but the testimony is unmistakable in its confirmation of the theory of the influence of the angle of the thread. It will be seen that every case having an angle above 12° 30′ was successful, and every case below 9° nnsuccessful, the overlapping of the successful and unsuccessful worms in the intervening region being what is to be expected in the bor-

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der region between good and bad practice. This band of uncertain results is in fact narrower than we would have any right to expect from a collection of data from miscellaneous sources, and could the inquiry be widened in scope the width of this band would doubtless be increased. As throwing light on these cases it should be remembered that case 16 is known to have been made successfnl only by careful attention to the material used, the first worms made having been failures, and that 3, which is near 16, and was a failure, had an excessive speed, while 11 at a lower angle, and a success, had a very moderate speed. At a higher speed 11 would probably have failed, and at a lower speed 3 would probably have been a success. It is believed that Fig. 3 points out clearly the nature of the worm problem and the conditions of success in its solution.

Some investigations of this subject have been made by C. Bach and E. Roser

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at the Royal Technical High School, Stuttgart, Germany, published in the "Zeitschrift des Vereines Deutscher Ingenieure," and republished in the "American Machinist" for July 16 and 23, 1903.

The worm was three-threaded, of steel, not hardened, having a total diameter of 76.6 mm. and a lead of 76.2 mm., giving a helix angle of 17° 34′. The worm wheel was of bronze with 30 teeth and a pitch diameter of 242.6 mm. The worm was immersed in an oil bath into which thermometers were inserted, and in the conduct of the experiments the apparatus was operated under various loads and speeds. Observations of the temperatures of the oil and surrounding air were made until the difference reached a constant value. Fig. 7*a* shows the relation of speed and pressure at two of the observed constant differences of temperature. In this diagram the temperatures are in degrees centigrade and the sliding velocities in meters per second.

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The curves will be seen to be distinctly parabolic in character.

In the notation of the experimenters:

P = pressure on teeth in kilograms

t = pitch of teeth in centimeters.

b = breadth of wheel measured on the arc at the roots of the teeth in centimeters.

k = a constant.

It is then assumed that

$$P = k\,b\,t.$$

The value of k is found from the equation:

$$k = c\,(t_o - t_e) + d,$$

in which

$$c = \frac{.0669}{v} + .4192.$$

$$d = \frac{109.1}{v + 2.75} - 24.92.$$

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t_o = temperature of the oil in degrees centigrade.

t_e = temperature of the surrounding air in degrees centigrade,

and in which again,

v = sliding velocity at the pitch line in meters per second.

It is obvious, also, that the diagram furnishes a means by which the load and speed at which a worm gear works satisfactorily being known, the proper load for other speeds may be ascertained.

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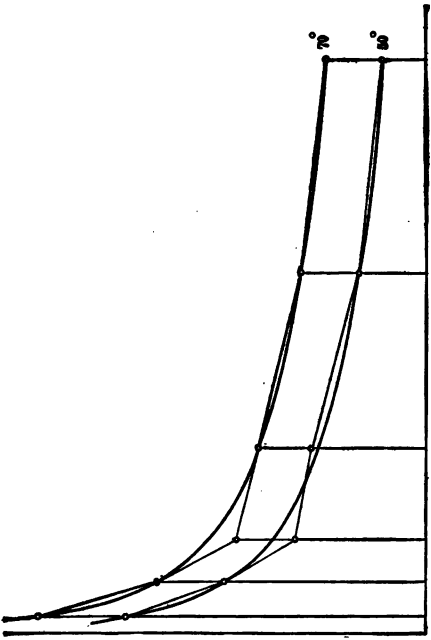

FIG. 7a.—RELATION OF PRESSURE AND VELOCITY.

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STEP BEARINGS.

The step bearings of worms have been a source of trouble alongside of the worm itself. It is obvious enough that the increase of the thread angle will relieve the step of wear as well as the worm thread. Expedients are possible with the step, however, which are not available with the worm. Ball and roller step bearings have been extensively tried, and while some have been successful with them, the general results are believed to have been unsatisfactory. The troubles from ball bearings arise from the tendency of the balls to break up under heavy loads and to score the pressure plates, while conical rollers, which geometrical considerations call for, have in some instances made trouble from their outward radial pressure cutting out the confining ring.* The multiple washer thrust bear-

* Mr. C. R. Pratt has had marked success with roller thrust bearings in which the rollers were short cylinders kept in position by a distance plate or cage having suitable openings arranged in spirals. See "The American Machinist," June 23, 1901.

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ing is used by many, and is undoubtedly entirely successful. Many of the readers of this volume have no doubt seen this pattern of bearing without reflecting upon the principle which lies at the bottom of it. When several loose washers are interposed between the shaft collar and the face of the shaft bearing, it is obvious that slipping may occur between any pair of faces, and that this slipping will take place between those surfaces which at the moment offer the least friction. Should these surfaces from any cause increase their resistance the slipping will be at once transferred to another joint, the various surfaces acting as mutual safety valves to one another, any surface which gets into the condition of incipient heating or cutting being at once relieved by another taking up the work. Fig. 6 shows one of these bearings as made by the Newton Machine Tool Works. The Newton Works formerly followed the usual practice and made the washers *a* alternately of hardened

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steel and bronze, but consider that they have improved on this by substituting white cast-iron for the steel. These castings are obtained from the malleable iron foundries, and are in fact unannealed malleable castings. Of course these castings cannot be machined, and they are therefore prepared for use by grinding on a cup-shaped emery wheel. They are dropped into a socket on the end of a shaft, which is revolved by hand, and are thus presented to the face of the cup-shaped wheel. This plan results in the grinding marks crossing the faces in all directions, instead of being in circles as in lathe-finished pieces. All mechanics understand the advantage of having the tool marks in a direction different from that of the motion of the parts, which advantage this method of construction secures. Another feature of the Newton bearings consists in making the holes in the washers larger than the shaft on which they are placed. This construction introduces an irregular compound mo-

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tion of the surfaces upon one another, the advantages of which are well understood. In the Newton washers the holes are $\frac{1}{32}$-inch larger than the shaft, though Mr. Newton considers that this might be increased with probably good results.

Three radial oil grooves are cast in each face of these disks, the use of which is obvious. The Hewes & Phillips step, at the right-hand bearing, will be seen to include three washers, while the left-hand step has lenticular-shaped disks—a construction which has also found favor elsewhere.

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PART II.

SPIRAL GEARING.

SPIRAL GEARS COMPARED WITH SPUR GEARS—THE DIFFERENT PITCHES.

SPIRAL gears are not to blame for the undoubted fact that they are somewhat troublesome to lay out, the difficulties of the problem being due to the limitations of workshop facilities and not to the geometrical nature of the gears themselves. It is easy to understand and explain the action of an existing pair of spiral gears. More than this it is easy to lay out a pair of such gears which shall exactly meet all the conditions of the case except one—they cannot, except through rare good luck, be made with the appliances at hand. To be more specific, the circum-

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ference cannot usually be divided into an exact whole number of teeth by any stock cutter, and the real problem becomes the readjusting of the diameters of the gears and the angle of the teeth, so that stock cutters shall make an exact whole number of teeth.

With spur gears it is only necessary to multiply the (circumferential) pitch of the cutter by the number of teeth to be cut to obtain the circumference of the gears. With spiral gears this operation gives the length of a portion of a spiral, or, more properly, helix, wound upon the pitch surface. We do not know the angle of this helix, the diameter of the pitch cylinder upon which it is wrapped or even what part of a complete turn the known portion comprises. The length is known for each gear and nothing more, and it becomes a matter of trial to find the diameters of the gears and the helix angle to suit this portion of the helix and at the same time fill the required center distance.

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Fig. 8 is a conventional representation of the pitch surface of a spiral gear, the surface being extended beyond the limits of the gear in order that the two helixes with which we are concerned may be shown. The first of these, *a b c d e f*, is the tooth helix and the second, *a g h d i p*, is the normal helix. The tooth helix is of importance because it defines the angle of the teeth. Given the diameter of the pitch surface, the helix may be defined by the angle *k a l* or by the length *a f*, in which it makes a complete turn—that is, by its pitch. For the determination of the speed ratio of a pair of gears the former method is the more convenient, but the tables supplied with universal milling machines which are used in setting up the machine employ the latter method.

In all spiral gear problems we have two pitches to deal with—the pitch of the tooth helix and the pitch of the teeth. The latter may be measured in several ways. First is the value *a n*, measured on the circumference or the *circumfer-*

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ential pitch, which is analogous to the pitch of spur gears; second is the value *a o* measured on the normal helix or the *normal pitch*, for which the cutters must be selected; third is the value *a r* measured parallel with the axis or the *axial pitch*.* Since the cutters must be selected with reference to the normal pitch the length of the normal helix is naturally of importance in connection with the number of teeth in the gear. The normal pitch multiplied by the number of teeth must naturally equal the length *a g h d* of this helix measured between its intersections *a* and *d* with the helix of a single tooth. Note that the length of the normal helix to be considered is the length *a g h d* between its intersections with the tooth helix and not the length

* Of two mating gears the circumferential pitch of one is equal to the axial pitch of the other, and *vice versa*. The axial and circumferential pitches are of small importance, except in the case of worm gearing. In making a worm we deal with axial pitch, and in making a worm wheel, with circumferential pitch. In such spiral gears as are made with a formed milling cutter (not a hob) we are concerned chiefly with normal pitch.

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a g h i p q of a complete turn around the cylinder. That this is true may be seen by reference to Fig. 9, in which the angle *k a l* is nearly a right angle. It is apparent from this illustration that the length of the normal helix from *a* to *d* takes in all the teeth and that *a o*, multiplied by the number of teeth, must equal *a h p d* and not *a h p q*. This length *a h p d* is always less than *a h p q*, and usually much less. Fig. 10 *A* is a development of Fig. 9 on a reduced scale, *a d* being the developed length of the normal helix. Fig. 10 *B* and Fig. 10 *C* show how with the same circumferential pitch and the same number of teeth but a reduced value of the angle *k a l*, the length of the normal helix which cuts all the teeth grows shorter until it may make but a small part of a complete turn around the cylinder. It is clear that in all cases the line *a d* cuts all the teeth precisely as does the circumference *a a*, which goes completely around the cylinder. It is also clear that if the normal pitch is de-

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cided upon at the start, a diameter of cylinder and a helix angle must be found such that the normal pitch, multiplied by the number of teeth, shall equal the length of the normal helix between two intersections with the tooth helix.

It is natural to ask: Why not employ the circumferential pitch and so deal directly with the circumference instead of the normal helix? Because we do not know what it is. The normal pitch is determined by the cutter used, while the circumferential pitch depends also upon the helix angle, and until this angle is known the circumferential pitch is not known.

In the extreme case of a spiral gear in which the helix angle is so small that the gear becomes a single thread worm, as in Fig. 11, points *o* and *d* coincide and the length of the helix between *a* and *d* becomes the normal pitch. It is, however, true as before that the normal pitch, multiplied by the number of teeth, which is now one, is still equal to the length of

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the normal helix between two intersections with the tooth helix.

A glance at Fig. 10 will show that in gears of the same diameter the length of the normal helix* grows shorter as the angle *k a l* grows less, and hence that it and its gear will contain successively fewer and fewer teeth of the same normal pitch. That is to say, the number of teeth in a gear varies with the helix angle as well as with the diameter and *the number of teeth in two gears of the same normal pitch is not necessarily proportional to the diameters.* In fact, it is never so proportional, except when the angle *k a l* is equal to 45 degrees. *The diametral pitch of the cutters and the diameter of the gear thus do not determine the number of teeth.*

The two facts thus developed are fundamental and will bear re-stating :

First, *The number of teeth is equal to*

* "Length of normal helix" is to be understood as meaning the length of that helix between two intersections with the same tooth helix.

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the length of the normal helix divided by the normal pitch.

Second, *The numbers of teeth in a pair of gears are not proportional to the diameters, except when the angle of the tooth helix is* 45 *degrees.*

THE SPEED RATIO.

Fig. 12 illustrates the simplest possible case of a pair of spiral gears. The gears are of equal size and the tooth helix has an angle of 45 degrees. Such a pair of gears will obviously run at the same speed —that is, have a speed ratio of 1—and as obviously both will have the same number of teeth. Now, unlike spur gears, there are two ways in which the speed ratio of such a pair of spiral gears may be varied. First, the diameters of the gears may be changed, as with spur gears, the angle of the tooth helix remaining unchanged, as in Fig. 13; and second, the angle of the helix may be changed, the diameters of the gears remaining unchanged, as in Fig. 14. These methods act in very dif-

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ferent ways. The first method is analogous to the procedure with spur gears. As with spur gears, the circumferential or pitch line speed of the two gears remains, as before the change, equal, but the length of the circumference of the two gears is unequal and the larger one thus has a less number of revolutions than the smaller one. The second method is entirely unlike anything seen in connection with spur gears. By it the pitch line speeds of the two gears are made unequal, and hence, while their diameters are equal, the lower one revolves the more slowly. This points out another fundamental difference between spiral and spur gears: With spiral gears, unless the helix angle is 45 degrees, *the pitch line speeds of two mating gears are not the same.*

The two methods of changing the speed ratio shown in Figs. 13 and 14 may be combined. That is, part of the desired change in speed may be obtained by changing the diameters of the gears and the remainder by changing the angle of

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the helix. Given the speed ratio and the diameter of one of the gears, we may assume a helix angle and find a diameter for the second gear to go with it which shall give the desired speed ratio and, having done this, a second angle may be assumed and a second diameter be found. There are thus an indefinite number of combinations of angles and diameters which will give the required speed ratio. Note, however, that with the diameter of one gear fixed, every change in the diameter of the other changes the distance between centers, that not every angle of helix can be obtained by the gears which are furnished with universal milling machines, and that if ready-made cutters are to be used, the lengths of both normal helixes must be exact multiples of the normal pitch of the teeth.

The limitation of the helix angle is not, however, as serious as is usually supposed. The tables for spirals which have heretofore been supplied with universal milling machines give but a few

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of the spirals which can be obtained with the change gears which are regularly supplied with the machines. In the case of the Brown and Sharpe universal milling machine, about two thousand spirals can be cut with these gears.

Geometrically speaking, there is a wide range of choice in the helix angle. As regards the desirability of different angles from the standpoint of durability, the conditions are essentially the same as in worm gearing. Reference to Part I, will show that the most favorable angle for durability is at about 45 degrees. There is, however, but a trifling increase in wear down to 30 degrees, no serious increase down to 20 degrees, and no destructive increase down to about 12 degrees. Where gears are to transmit considerable power the best results should attend the use of angles between 30 and 45 degrees, while angles as low as 20 degrees may be used without hesitation, and as low as 12 degrees if the gears are

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to run in an oil bath or do light work only. The angle may also be increased above 45 degrees by similar amounts and with similar results.

Fig. 15 is a development of the gears of Fig. 14, the angle α of Fig. 15 being equal to $k\,a\,l$ of Fig. 14, but in reversed position, because in Fig. 14 the upper side of the driver is seen, while in Fig. 15 the direction of the teeth is that of the lower side of the driver.

It is clear that if the driver move in the direction of the arrow it will, while moving the distance $a\,b$, push the driven gear the distance $b\,c$, and the pitch line speeds will have the relation :

$$\frac{\text{p.l. speed follower}}{\text{p.l. speed driver}} = \frac{b\,c}{b\,a} = \tan\,\alpha.$$

If the gears have the same diameters, their number of revolutions will be in the same ratio as their pitch line speeds—that is :

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$$\frac{\text{rev. follower}}{\text{rev. driver}} = \tan \alpha,$$

or

$$\text{rev. follower} = \text{rev. driver} \times \tan \alpha$$

If the diameter of the follower be increased, its number of revolutions will be reduced in the same ratio—that is :

$$\text{rev. follower} = \frac{\text{diam. driver}}{\text{diam. follower}} \times \text{rev. driver} \times \tan \alpha,$$

or

$$\frac{\text{rev. follower}}{\text{rev. driver}} = \frac{\text{diam. driver}}{\text{diam. follower}} \times \tan. \alpha,$$

the angle α being taken from the driver.

This is a complete formula for the speed ratio of spiral gears having shafts at right angles. But for the limitations imposed by the use of stock cutters it, together with the fact that the sum of the diameters of the gears must equal twice the center distance, would be all that is required for designing such gears. Note that it differs from the corresponding

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formula for spur gears only by the introduction of the factor tan α.

THE PRELIMINARY SOLUTION.

The simple formula for the speed given above will be needed repeatedly, and had best be put in algebraic form.

Let r_1 = revolutions of driver,
r_2 = revolutions of follower,
d_1 = diameter of driver,
d_2 = diameter of follower,
α = helix angle of driver.

Then this formula becomes :

$$\frac{r_2}{r_1} = \frac{d_1}{d_2} \tan \alpha \qquad (1)$$

In any actual case the speeds are given and the diameters and helix angle must be found. We may assume a ratio for the diameters and find the angle, or we may assume an angle and find the ratio of diameters. It is desirable to assume the angle first, as on it depends, largely, the durability of the gears. To do this

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the above formula may be more conveniently written :

$$\frac{d_2}{d_1} = \frac{r_1}{r_2} \tan \alpha \qquad (2)$$

The sum of the diameters must equal twice the center distance, which we may call C. That is :

$$d_1 + d_2 = 2C,$$

or

$$d_2 = 2C - d_1.$$

Substituting this value for d_2 in (2) we obtain :

$$\frac{2\ C - d_1}{d_1} = \frac{r_1}{r_2} \tan \alpha$$

which, solved for d_1, becomes :

$$d_1 = \frac{2C}{\frac{r_1}{r_2} \tan \alpha + 1} \qquad (3)$$

Having assumed a value for α and substituted its tangent and the ratio of the desired speeds in (3), we find a value for d_1, and, having found d_1, d_2 may obviously be found by subtracting d_1 from $2C$.

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Such a solution is complete in a geometrical sense, and if it were feasible to make a cutter to suit each case, it would be complete in a practical sense also. When, however, we go a step further and find the length of the normal helixes, the probabilities are all against their being exact multiples of the pitch of any stock cutter. The solution so obtained must therefore be considered as provisional and be modified to suit the cutters to be used.

The table at the end of the book is republished here from "A Treatise on Milling Machines," by the courtesy of the Brown and Sharpe Manufacturing Company, and gives by inspection all the spirals which can be obtained by the gears supplied with the universal milling machines provided by them. Figs. 24 and 25 show the same make of machine with the vertical spindle milling attachment applied to the cutting of spirals which cannot be cut in the usual manner because of the interference of the work-table with the column of the machine.

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THE LENGTHS OF THE NORMAL HELIXES.

Fig. 16 is the development of a pair of gears placed in the most convenient position for showing the lengths of the normal helixes. The tooth and normal helixes are extended beyond the face of the gears. Let

c_1 = circumference of driver,
c_2 = circumference of follower,
d_1 = diameter of driver,
d_2 = diameter of follower,
l_1 = length of normal helix of driver between intersections with tooth helix,
l_2 = length of normal helix of follower between intersections with tooth helix,
α = tooth helix angle of driver.

Obviously

$$l_1 = c_1 \sin \alpha,$$
$$= \pi d_1 \sin \alpha,$$

or

$$\frac{l_1}{\pi} = d_1 \sin \alpha \qquad (4)$$

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$$l_2 = c_2 \cos \alpha,$$
$$= \pi d_2 \cos \alpha,$$

or
$$\frac{l_2}{\pi} = d_2 \cos \alpha \qquad (5)$$

Note that (4) and (5) give the lengths of the normal helixes divided by π and not their actual lengths. This is done because, in dealing with diametral pitch cutters the calculations are made less laborious. Dividing (4) by (5) gives :

$$\frac{l_1}{l_2} = \frac{d_1 \sin \alpha}{d_2 \cos \alpha}$$
$$= \frac{d_1}{d_2} \tan \alpha \qquad (6)$$

Comparing (1) with (6) proves what is almost self-evident, that the lengths of the normal helixes are to each other inversely as the number of revolutions, and hence that *a pitch which will exactly divide the short helix will also divide the long one* and that *the numbers of teeth in the gears are inversely as the speeds.*

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A PRACTICAL EXAMPLE.

An example will best illustrate the actual procedure. Thus assume the conditions of a pair of gears to be :

$$\frac{\text{revolutions of follower}}{\text{revolutions of driver}} = \frac{r_2}{r_1} = \frac{1}{4}$$

and

$$\text{center distance} = C = 4\tfrac{15}{32} = 4.468 \text{ inches.}$$

Assume further that a helix angle of 45 degrees would lead to a diameter for the driver which would make it too small for its shaft. We are, at the start, entirely at sea regarding the whole matter; but as an angle of 30 degrees is favorable to durability we may use it as a trial angle and see what it will lead to. Finding the tangent of 30 degrees in a table and substituting it and the value of $\frac{r_1}{r_2}$ in (3) we obtain :

$$d_1 = \frac{2 \times 4.468}{4 \times .57735 + 1}$$
$$= 2.7$$

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and

$$d_2 = 2 \times 4.468 - 2.7$$
$$= 6.236.$$

From (4) we find

$$\frac{l_1}{\pi} = 2.7 \times .5$$
$$= 1.35$$

and from (5)

$$\frac{l_2}{\pi} = 6.236 \times .866$$
$$= 5.4.$$

These values of d_1, d_2, $\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$ are the provisional values belonging with 30 degrees for α.

Assume next that it is desired to use cutters of 6 diametral pitch, the circumferential pitch of which is $\frac{\pi}{6} = .5236$. We may find the number of teeth which the normal helixes will contain by dividing their lengths by this circumferential pitch, but

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$$l \div \frac{\pi}{6} = \frac{l}{\pi} \times 6.$$

That is, the number of teeth of 6 diametral pitch which the provisional normal helixes will contain may be found by multiplying $\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$ respectively by 6. Performing this operation we obtain:

$$\frac{l_1}{\pi} \times 6 = 1.35 \times 6$$
$$= 8.1$$

and
$$\frac{l_2}{\pi} \times 6 = 5.4 \times 6$$
$$= 32.4.$$

The provisional normal helixes thus contain 8.1 and 32.4 teeth of the desired pitch, and as these numbers are impossible, we take the nearest whole numbers having the desired ratio of 1 to 4, namely, 8 and 32. That is, we decide to shorten the normal helixes until they contain exactly 8 and 32 teeth.

The meaning of this is shown graphi-

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cally in Fig. 17. Laying down $a\,b$ and $a\,c$ to represent the circumferences as found above and drawing the normal helixes at an angle of 30 degrees, we have found that the normal helix $a\,d$ of the driver will contain 8 teeth and a little more, and the normal helix $a\,e$ of the follower 32 teeth and a little more. As we cannot have a fraction of a tooth, we decide to cut off the ends of the helixes, making their lengths $a\,d'$ and $a\,e'$.*

FINAL SOLUTION BY CHANGING THE CENTER DISTANCE.

The most obvious way of carrying this out is simply to reduce the diameters of both gears, so as to make their circumferences $a\,b'$ and $a\,c'$ instead of $a\,b$ and $a\,c$. This change obviously reduces the center distance, but at this drawing-board

* Should the helixes come out a little short instead of a little long, they would be lengthened instead of shortened. Had the helix of the driver been of a length to contain, say, 8.3 teeth, that of the follower would obviously have contained 33.2 teeth, and, in the correction, more than an entire tooth would have been cut from it.

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stage of affairs this can often be done, and when it can be done it is the readiest way out of the difficulty. To determine how much to reduce the diameters we may first find the reduced lengths of the normal helixes, by dividing the final numbers of teeth—8 and 32—by 6 and then find the new diameters, or more simply, we may note, what is apparent from Fig. 17, that the lengths of the helixes and of the diameters have been changed in the same ratio as the numbers of teeth.

That is :

$$\frac{\text{final diameter}}{\text{provisional diameter}} = \frac{8}{8.1}$$

or

$$\text{final diameter} = \text{provisional diam.} \times \frac{8}{8.1}$$

That is :

$$\text{final } d_1 = 2.7 \times \frac{8}{8.1}$$
$$= 2.667$$

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and

$$\text{final } d_2 = 6.236 \times \frac{8}{8.1}$$

$$= 6.159$$

and $d_1 + d_2 = 2.667 + 6.159 = 8.825 =$ twice the new center distance.

GRAPHICAL SOLUTION WITH CHANGED CENTER DISTANCE.

All these determinations may be made graphically as in Fig. 18. Lay off $a\,b = 2\,C = 8\frac{15}{16}$ inches. At any convenient distance lay off the indefinite line $c\,d$ parallel to $a\,b$. At c lay off the provisional angle $\alpha = 30$ degrees. Draw $e\,f$ at any convenient point perpendicular to $c\,d$. Take $e\,f$ in the dividers and step it off from e toward d as many times as will represent the ratio of the desired speed of the driver divided by that of the follower. That is, in the present case, lay off $e\,f$ 4 times above e and thus obtain d. Draw $c\,a$ and $d\,b$ and extend them till they meet

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at g.* Draw $g\,e$, giving $a\,h$ and $b\,h$, which are provisional diameters of driver and follower respectively. Draw $h\,p$ perpendicular to $a\,b$, at h lay down $h\,k$ and $h\,l$ to repeat α, and from h strike arcs $a\,k$ and $b\,l$. Draw $k\,o$ and $n\,l$ perpendicular to $a\,b$, and we have the provisional values,

$$a\,h = d_1$$
$$b\,h = d_2$$
$$h\,o = \frac{l_1}{\pi}$$
$$h\,n = \frac{l_2}{\pi}$$

Scale $h\,o$ and $h\,n$ and multiply them by the diametral pitch number—6. If the results are not whole numbers, as they usually are not, select the nearest whole numbers having the desired speed ratio, and they are the final numbers of teeth. Divide these numbers by the diametral pitch number to obtain the final values

* Had $c\,d$ been taken shorter than $a\,b$, g would have fallen to the left of the diagram, but the construction would otherwise have been unchanged.

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of $\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$ and lay them down as $h\,o'$ and $h\,n'$. Draw $o'\,k'$ and $l'\,n'$ and $k'\,a'$ and $l'\,b'$, giving:

$a'\,h$ = the final d_1,
$b'\,h$ = the final d_2,
$a'\,b'$ = twice the new center distance.

FINAL SOLUTION WITH UNCHANGED CENTER DISTANCE.

If the center distance cannot be varied, the condition of things is shown in Fig. 19. Having found that the provisional normal helixes will not contain an exact number of teeth, and cut them off and obtained the smaller circumferences $a\,b'$ and $a\,c'$ of Fig. 17, the problem becomes to find a new value of α, indicated in Fig. 19, as α', which shall give such diameters as to restore the old center distance. That is, $a\,b'' + a\,c''$ of Fig. 19 must be equal to the original $a\,b + a\,c$ of Fig. 17. It will be seen by a glance at Fig. 19 that the reduced value of α in-

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creases the diameter of the driver, but reduces that of the follower.

It is probable that the correct angle and diameters for this case can be found only by trial. From (4) we have :

$$d_1 = \frac{l_1}{\pi \sin \alpha} \qquad (7)$$

and from (5)

$$d_2 = \frac{l_2}{\pi \cos \alpha} \qquad (8)$$

We also have :

$$d_1 + d_2 = 2\,C;$$

that is $$\frac{l_1}{\pi \sin \alpha} + \frac{l_2}{\pi \cos \alpha} = 2\,C,$$

which easily reduces to :

$$1 + \frac{l_2}{l_1} \tan \alpha = \frac{2\,C}{\frac{l_1}{\pi}} \sin \alpha \qquad (9)$$

To solve this equation we know that $\frac{l_2}{l_1}$ = the speed ratio = 4 and in a preceding section we found that the cor-

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rected value of $\frac{l_1}{\pi}$ bears the same relation to the provisional value that the corrected number of teeth bears to the provisional number. That is, the corrected value of $\frac{l_1}{\pi} = 1.35 \times \frac{8}{8.1} = 1.33$. Substituting these values and the value of C, the only unknown quantity is α, but, from the form of the equation, α can be found only by trial and error, the result of each trial being a closer and closer approximation to the truth. The operation is simple enough, but it involves the repeated multiplication of decimals. For the earlier steps a slide rule will greatly abbreviate the work, while with a Sexton's omnimeter an accuracy can be obtained sufficient for all the steps and a piece of drudgery be converted into an exhilarating chase. The results which follow were obtained with an omnimeter. Trying the provisional angle of 30 degrees in order to note the result and making the substitutions in (9) we obtain :

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$$1 + 4 \times .577 = 6.702 \times .5$$

or $$3.308 = 3.351.$$

This is not correct, as we knew it would not be. The right-hand number is larger than the left-hand number, which will always be the case if our trial value of α is too large, and *vice versa* if it is too small. Trying 28 degrees we obtain :

$$1 + 4 \times .532 = 6.702 \times .469$$

or $$3.13 = 3.145.$$

The right-hand number is still too large, showing 28 degrees to be too large. Trying 26 degrees we obtain :

$$1 + 4 \times .488 = 6.702 \times .438$$

or $$2.953 = 2.935.$$

The right-hand number is now too small, showing that 26 degrees is too small. Trying 26 degrees 30 minutes we obtain :

$$1 + 4 \times .499 = 6.702 \times .446$$

or $$3. = 2.99$$

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This angle is still too small, though obviously very nearly correct. Trying 26 degrees and 40 minutes we obtain :

$$1 + 4 \times .502 = 6.702 \times .449$$

or $$3.01 = 3.01$$

Within the limits of accuracy of the omnimeter this is correct, and as that instrument shows the effect of so small a change in the angle as 10 minutes the result is obviously close enough. Having found the angle we are now in shape to find the final diameters. Substituting in (7) and (8) we obtain :

$$d_1 = \frac{1.333}{.4488}$$
$$= 2.971$$

and

$$d_2 = \frac{5.333}{.8936}$$
$$= 5.968$$

and $5.968 + 2.971 = 8.939$, which is twice the required center distance as closely as can be expected.

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GRAPHICAL SOLUTION WITH UNCHANGED CENTER DISTANCE.

These determinations may also be made graphically, as shown in Fig. 20, which repeats the construction of Fig. 18 up to the finding of the provisional values of $a\,h$ and $b\,h$.

In Fig. 18 h was a fixed point, while a and b were not. In the present case a and b are fixed, but h is not, and to find $\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$ in the present case it is better to lay off α at a and b instead of at h, as was done in Fig. 18. Therefore lay down α at a and b, as shown, and strike arcs $h\,k$ and $h\,l$. Draw $k\,o$ and $l\,n$, giving the provisional values :

$$a\,o = \frac{l_1}{\pi}$$

$$b\,n = \frac{l_2}{\pi}$$

Measure these distances and multiply them by the diametral pitch number—in this case 6—to find the numbers of teeth

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going with the assumed value of α. If, as is usually the case, the products are not whole numbers, take the nearest whole numbers having the ratio of the desired speeds, divide them by the diametral pitch number and obtain the corrected values of $\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$. Lay off $a\,o'$ and $b\,n'$ equal to these corrected values. Assume a new trial angle, indicated as α', and lay it down at a and b. Draw $o'\,k'$ and $n'\,l'$, and through the intersections k' and l' draw the dotted arcs. If these arcs meet in tangency at h', then we have the final values :

$$a\,h' = d_1$$

and

$$b\,h' = d_2$$

If the arcs do not thus meet in tangency, assume another value for α and try again.

Graphical methods are sufficiently accurate for many purposes, especially if laid out upon an enlarged scale, and if it is remembered that if the values used for

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$\frac{l_1}{\pi}$ and $\frac{l_2}{\pi}$ are a little short of the true values, the shortage will be divided among the teeth with the only result of a little slackness in the teeth.

FINDING THE PITCH OF THE TOOTH HELIX.

It now only remains to find the pitch of the tooth helixes in order to set up the machine. In Fig. 16 let

p_1 = pitch of tooth helix of driver,
p_2 = pitch of tooth helix of follower;

then $$p_1 = c_1 \tan \alpha,$$
$$= \pi d_1 \tan \alpha,$$

and $$p_2 = c_2 \cot \alpha,$$
$$= \pi d_2 \cot \alpha.$$

Inserting the final values of d_1, d_2, and α we obtain :

$$p_1 = \pi \times 2.971 \times .502$$
$$= 4.684,$$

and $$p_2 = \pi \times 5.968 \times 1.991$$
$$= 37.33.$$

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These values may be found graphically as in Fig. 21. Lay down α and make

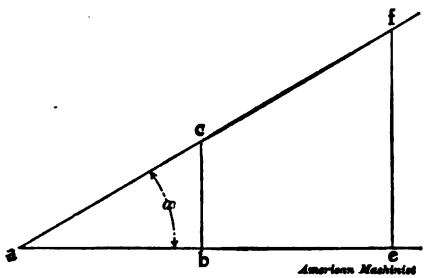

FIG. 21.—FINDING THE PITCH OF THE TOOTH HELIX.

$ab = \pi d_1$ when $bc = p_1$. Find $ef = \pi d_2$ and $ae = p_2$.

SPECIAL SOLUTION FOR A HELIX ANGLE OF 45 DEGREES.

Remembering that $\frac{l_2}{l_1} = \frac{r_1}{r_2}$, the latter fraction may be substituted for the former in (9) together with the assumed value of α and the provisional value of $\frac{l_1}{\pi}$

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be thus found without first finding the provisional values of d_1 and d_2. The final values of d_1 and d_2 may then be found from (7) and (8). Such a solution is in some respects neater than the one given, but it is believed to be less easily followed and comprehended, and, moreover, it does not, for the general case, furnish the basis of so good a graphical solution. This method is at its best in the case of a helix angle of 45 degrees, although this angle does not differ from any other in requiring a change in the center distance if it is to be adhered to. In the nature of the case, if we deny ourselves the liberty of changing the angle, the only recourse is to correct the length of the helix by changing the diameters. As this angle is used more than any other—perhaps more than all others—this solution for it is given.

Remembering that in (9) $\frac{l_2}{l_1} = \frac{r_1}{r_2}$, that $\tan 45° = 1$, and that $\sin 45° = .7071$,

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that equation becomes *for this angle only:*

$$1 + \frac{r_4}{r_2} = \frac{1.4142\ C}{\frac{l_1}{\pi}}$$

which may be made to read :

$$\frac{l_1}{\pi} = \frac{1.4142\ C}{1 + \frac{r_1}{r_2}} \qquad (10)$$

Inserting sin 45° = .7071 in (7) we have :

$$d_1 = \frac{\frac{l_1}{\pi}}{.7071} \qquad (11)$$

Equations (10) and (11) are all that are needed for solving any case having this helix angle.

Taking up the preceding problem again and solving it for this angle, we have from (10) for the provisional value :

$$\frac{l_1}{\pi} = \frac{1.4142 \times 4.468}{1 + 4}$$
$$= 1.2637.$$

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Multiplying this by the diametral pitch of the cutter to be used, we obtain :

$$1.2637 \times 6 = 7.58$$

as the provisional number of teeth in the driver. As it is an impossible number, we select 8 as the number of teeth and, dividing it by 6, we obtain :

$$\frac{8}{6} = 1.333$$

as the final value of $\frac{l_1}{\pi}$. Substituting this in (11), we obtain :

$$d_1 = \frac{1.333}{.7071}$$
$$= 1.885$$

as the final value of the diameter of the driver. The diameter and number of teeth of the follower for an angle of 45 degrees are found by multiplying the diameter and number of teeth of the driver by the speed ratio. That is :

$$d_2 = 1.885 \times 4$$
$$= 7.540$$

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and

number of teeth in follower $= 8 \times 4$
$= 32.$

Finally

$d_1 + d_2 = 1.885 + 7.54$
$= 9.425$
$=$ twice the final center distance.

GRAPHICAL SOLUTION FOR A HELIX ANGLE OF 45 DEGREES.

The graphical solution for this angle is simplified still more than the analytical one.

In case the adjustment to suit stock cutters is to be made by changing the diameters of the gears, that adjustment can be made as in Fig. 22, which repeats essentially, Fig. 17. Laying down the provisional values of the diameters at $a\,b$ and $a\,c$ and drawing $d\,e$ at the helix angle with the base line, the perpendiculars $b\,d$ and $c\,e$ to the normal helix line determine $a\,d$ and $a\,e$, the provisional lengths of the normal helixes divided by π.

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Changing these lengths to $a\,d'$ and $a\,e'$, such that when multiplied by the diametral pitch number of the cutter the results are whole numbers in the ratio of the required speeds, and drawing $d'\,b'$ and $e'\,c'$ gives $a\,b'$ and $a\,c'$, the corrected diameters.

If $\alpha = 45$ degrees the helix line becomes $f\,g$ and, with $b\,h$ perpendicular to it, $a\,b$ becomes equal to $a\,h$. In other words, instead of laying off the diameter lines at right angles to one another, they may, in this case, for purposes of determination, be laid off upon a straight line. This fact, in connection with the further fact that with a tooth angle of 45 degrees the diameters of the gears are inversely as the speeds, leads directly to the simple diagram given in Fig. 23, which is all that is required for determining graphically any pair of spiral gears having this tooth angle.

In Fig. 23 lay down $a\,b$ equal to twice the provisional center distance. Divide it at c into two parts the lengths of which

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are to each other in the ratio of the speeds. Draw de at an angle of 45 degrees with ab and draw ad and be at right angles with de, giving cd and ce the provisional lengths of the normal helixes divided by π. Multiply these lengths by the diametral pitch number. If the results are not whole numbers take the nearest whole numbers having the ratio of the desired speeds, which are, respectively, the numbers of teeth in the gears. Divide these numbers by the diametral pitch numbers and lay down cd' and ce' equal in inches of length to the quotients. Draw $d'a'$ and $e'b'$ perpendicular to de, and we have :

ca' = final diameter of one gear,
cb' = final diameter of the other gear,
$a'b'$ = twice the final center distance.

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SPECIAL SOLUTION FOR GEARS OF EQUAL DIAMETERS.

As was pointed out on page 51, the speed ratio of 1 which goes with gears of equal diameters and a helix angle of 45 degrees, may be varied by changing the diameters, the angle, or both. The case of gears having a helix angle of 45 degrees is therefore a special and simple case in which the desired speed ratio is obtained by recourse to a change in diameters only. An antithetical special and simple case is that in which the diameters of the two gears are made equal and the desired speed ratio is obtained by a change in the helix angle.

If this construction does not give rise to a smaller helix angle than that which is now well established as conducive to durability, there is no reason why it should not be used. This angle, it will be remembered, is about 12 degrees, which may be safely used for light work and probably for heavy work if the gears

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are to run in an oil bath. Under less favorable circumstances it is not desirable to go much below 20 degrees. With gears of equal diameters these angles correspond to speed ratios of about $4\frac{3}{4}$ and $2\frac{1}{4}$ to 1, respectively, and it hence follows that for these two cases and for speed reductions not greater than these, equal gears may be freely used.

Going back to the notation of pages 57 and 60, and letting p = the diametral pitch of the cutter to be used, formula (1), page 57, becomes for gears of equal diameters in which $d_1 = d_2$

$$\frac{r_2}{r_1} = \tan \alpha. \qquad (12)$$

It has also been shown that

$$\frac{l_1}{\pi} = d \sin. \alpha$$

that

$$\frac{l_2}{\pi} = d \cos. \alpha$$

and that the value of $\frac{l}{\pi}$ for either gear multiplied by the diametral pitch equals the number of teeth in that gear. That is:

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$p\,d$ sin. α = number of teeth in driver, and (13)

$p\,d$ cos. α = number of teeth in follower. (14)

If we are to use gears of equal diameters we are of course limited to tooth or helix angles appropriate to equal diamters, and as we have denied to ourselves the right to adjust the length of the normal helix by adjusting the ratio of the diameters and the helix angles, we can only do it by changing both diameters together. That is, our assumed diameters and center distance are provisional only and will ordinarily have to be changed in order to secure an exact whole number of teeth.

Practical Example: Required a pair of gears of approximately 5 inches diamter, the driver to run four times as fast as the follower; that is $\frac{r_2}{r_1} = \frac{1}{4}$.

Substituting in (12) we obtain at once tan. $\alpha = .25$, and referring to a table of natural tangents, we find that $\alpha = 14° 2'$.

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Referring to the sine and cosine columns of the table, we find that sin. $\alpha = .2425$ and cos. $\alpha = .9701$.

Assuming, say, a diametral pitch, and substituting these values in (13) and (14) we obtain for the provisional number of teeth in the driver:

$$8 \times 5 \times .2425 = 9.7$$

and in the follower:

$$8 \times 5 \times .9701 = 38.8.$$

These numbers being impossible, because fractional, we select the nearest whole numbers of teeth having the desired ratio of 4, namely, 10 and 40, and have now to find the corrected diameters to go with these tooth numbers. To do this it is only necessary to remember that the diameters are changed in the same ratio as the numbers of teeth; that is:

$$\text{Corrected diameter} = 5 \times \frac{10}{9.7}$$

or,

$$5 \times \frac{40}{38.8} = 5.155.$$

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GRAPHICAL SOLUTION FOR GEARS OF EQUAL DIAMETERS.

The graphical solution is shown in Fig. 26. Lay down $a\,b$ equal to the assumed or provisional diameter. Divide it into as many parts as will represent the speed ratio—in this case 4—and draw the perpendicular $b\,c$ equal to one of these parts. Draw $a\,c$ and $b\,a\,c$ is the value of α sought. Extend $b\,c$ and repeat α as shown. Strike the arc $a\,d$ and draw $d\,e$ perpendicular to $a\,b$, giving $b\,e = d$ sin. α of (13). Scale $b\,e$ and multiply it by the diametral pitch number 8, obtaining the provisional number of teeth in the driver, 9.7. Increase this to 10 and divide it by the diametral pitch number 8, giving 1.25, the corrected value of d sin. α. Lay this down from b, giving e'. Draw $e'\,d'$ and $d'\,a'$, giving the corrected diameter $a'\,b$. The corrected number of teeth in the follower is of course four times that in the driver, and the helix angle of the follower is the complement of that of the driver.

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THE SELECTION OF THE CUTTER.*

There is but one more question which may be mentioned, and that is the selection of the cutter for the spiral gear. If we consider a helix drawn upon a cylinder and then imagine a plane cutting that cylinder normal to the helix at any point, the plane will cut from the cylinder an ellipse having one extremity of its minor axis at the point where the helix is cut by the plane. The minor axis of this ellipse will be $2R$ and the major axis $\frac{2R}{\cos \alpha}$. If we cut a spiral gear in the same way at a point midway between two teeth, the form of the space shown will be the true normal shape which the cutter must be, and the above ellipse will be cut from the pitch cylinder. Now the curvature of this normal section of the gear at the point indicated must be the same as the curvature of the eclipse at the extremity

* By J. N. Le Conte in the "American Machinist." The helix angle used by Professor Le Conte is the complement of the angle heretofore used in his book.

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of its minor axis. If we call the major and minor axes of an ellipse $2a$ and $2b$, respectively, the radius of curvature at the extremity of the minor axis is $\rho_0 = \frac{a^2}{b}$.

But in our case $a = \frac{R}{\cos \alpha}$, and $b = R$,

Hence: $$\rho_0 = \frac{R}{\cos^2 \alpha}.$$

This is the *radius* of what might be called the "oscillating spur gear." The number of teeth on this gear will be:

$$N_0 = 2 P \rho_0 = \frac{2 P R}{\cos^2 \alpha}.$$

But on the spiral gear itself we have:

$$N = 2 P R \cos \alpha$$

Hence: $$N_0 = \frac{N}{\cos^3 \alpha}$$

and not $\frac{N}{\cos^2 \alpha}$ as sometimes stated.

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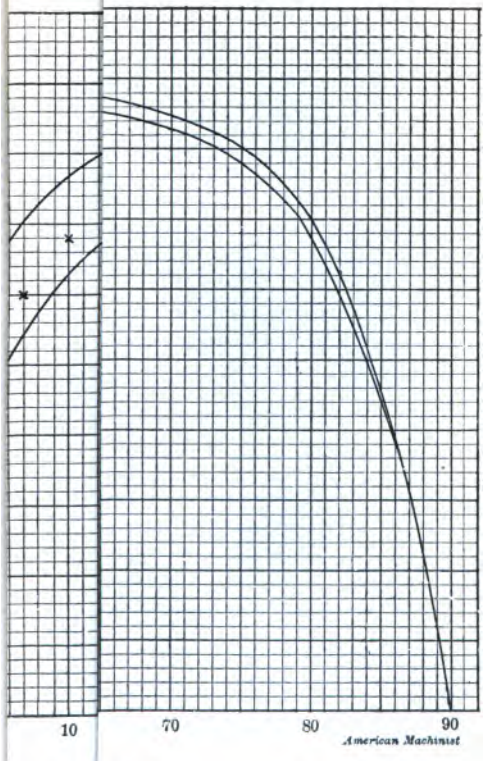

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rs connecting crosses indicate the same worm at different speeds
ce of crosses above 200 foot line has no quantitative significance.

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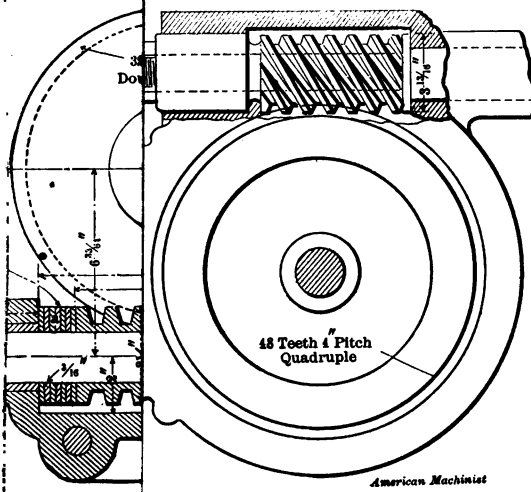

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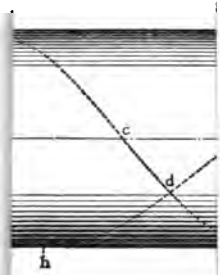

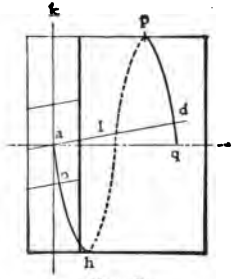

FIG. 9.

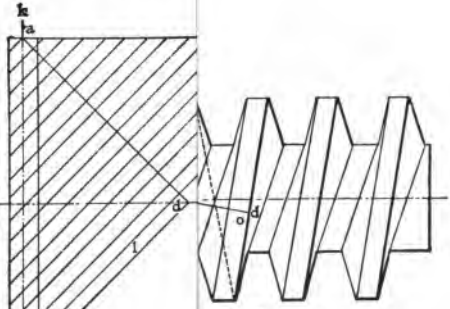

FIG. 11.

FIG. 10.

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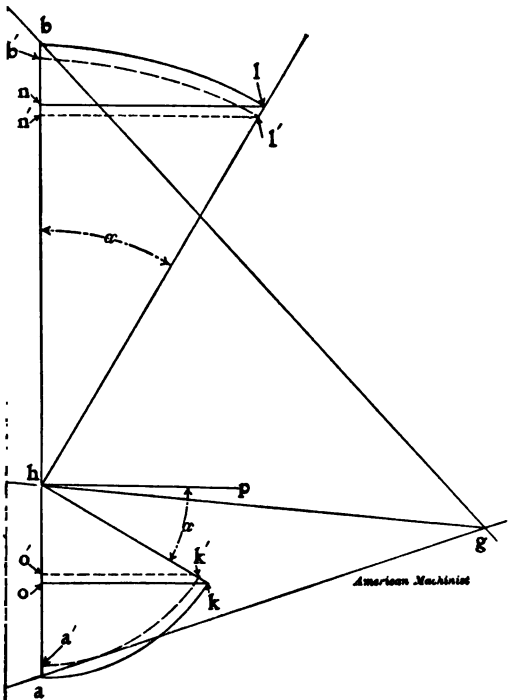

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